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ACOUSTIC INSTABILITIES IN SYNGAS FIRED COMBUSTION CHAMBERS

Sikke A. Klein and Jim B.W. Kok

Department of Mechanical Engineering
University of Twente
P.O. Box 217, NL 7500 AE
Enschede, The Netherlands

ABSTRACT

Gas turbines fired on syngas may show thermo-acoustic combustion instabilities. The theory on these instabilities is well developed. From this theory it can be shown that the acoustic system of a combustion installation can be described as a control loop with a set of transfer functions. The transfer function of the flame plays a decisive role in the occurrence of combustion instabilities. It is however very difficult to predict this flame transfer function analytically. In this paper a numerical method will be presented to calculate the flame transfer function from time-dependent combustion calculations. Also an experimental method will be discussed to determine this flame transfer function. Experiments have been performed in a 25 kW atmospheric test rig. Also calculations have been done for this situation. The agreement between the measurements and CFD calculations is good, especially for the phase at higher frequencies. This opens the way to apply CFD-modeling for acoustics in a real gas turbine situation.

NOMENCLATURE

A	Area [m^2]
$\tilde{b}_2 \ldots \tilde{b}_4$	Transfer function between 1D sound waves [-]
$\tilde{H}_{meas}$	Measured transfer function [-]
Q	Heat release [J/s]
R	Amplification factor [W.m^3/s^3]
$Refl$	Acoustic reflection factor [-]
$\tilde{T}$	Transfer function
T	Temperature [K]
U	Mean velocity [m/s]
V	Volume [m^3]
c	Speed of sound [m/s]
f	Mixture fraction [-]
	Frequency [Hz]
g	Variance of mixture fraction [-]
k	Wave number [1/m]
	Turbulent kinetic energy [m^2/s^2]
l	Length [m]
p	Acoustic pressure [Pa]
q	Local heat release [J/(s.m^3)]
t	Time [s]
u	Acoustic velocity [m/s]
U	Mean velocity [m/s]
x	Axial co-ordinate [m]

Greek symbols

γ	Poisson constant [-]
ε	Turbulent dissipation [m^2/s^3]
φ	Phase
τ	Time delay [s]
ρ	Density [kg/m^3]
ω	Frequency [rad/s]

Subscripts

$_{bur}$	Burner
$_{cc}$	Combustion chamber
$_{n}$	Natural

Superscripts

$'$	Fluctuation in time domain
$\tilde{}$	Frequency domain variable

INTRODUCTION

A large problem in gas turbine combustion is the occurrence of thermo-acoustic instabilities. Especially lean premixed burners may show an unstable behavior. But also gas turbines fired on (low calorific) coal gas have shown these instabilities as reported by Scalzo et al. in a Dow chemical plant (1990) and Stambler (1996) in the Buggenum power plant in the Netherlands.

There has already been a lot of research on flame instabilities. The focus of the research during the last decade has mainly been on premixed natural gas flames, both laminar and turbulent (see e.g. Poinsot (1987)). Little research has been performed on turbulent non-

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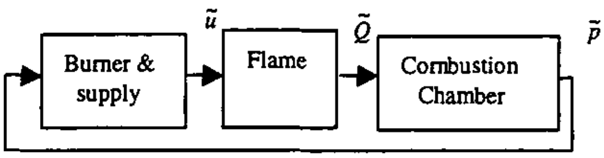

Figure 1: The control loop of the acoustic system of a combustion installation.

premixed flames during the last decade, Bohn and co-workers had some publications on this subject (see e.g. Bohn (1996)).

As an active component the flame has a major effect on the occurrence of thermo-acoustic instabilities. This can be described with a flame transfer function in the frequency domain. Due to the complex interaction of fluid mechanics, chemistry and thermodynamics it is not possible to calculate this flame transfer function analytically. For simple flames it can be measured but for a real gas turbine flame this will not be possible. Therefore a method based on time-dependent computational fluid dynamic calculations (CFD) has been developed to calculate this flame transfer function. The CFD calculations presented in this paper have been performed with the commercial CFD-package CFX in which the present model has been implemented. In this paper this CFD-method will be discussed and the results of it are compared with measurements performed in an atmospheric test rig.

The contents of the paper

In the first section of this paper the background theory of combustion driven oscillations will be discussed. Then the experimental method to measure the acoustic behavior of a flame is described. In the next section a method will be presented to determine the acoustic amplification by a flame from CFD calculations. The experimental results and the CFD results are compared with each other. From these results some conclusions are drawn and the future developments are given.

THEORY

A flame can be a strong volume source of sound. This is due to the large volumetric expansion in a flame. If this expansion fluctuates in time, the flame will generate sound. This expansion, which is a result of the heat release by the flame, appears as a monopole source term in the wave equation (the influence of the mean flow on the acoustic propagation is neglected):

$$\frac{1}{c^2}\frac{\partial^2 p'}{\partial t^2} - \nabla^2 p' = -\frac{\gamma-1}{c^2}\frac{\partial q'}{\partial t} \tag{1}$$

The fluctuating heat release (q') can have several origins:

1. The flame acts as an autonomous source of sound; the fluctuating heat release is caused by turbulent fluctuations.
2. The flame as amplifier of sound: The heat release by the flame fluctuates with the acoustic pressure in the combustion chamber. In this case the flame amplifies (or damps) the acoustic pressure.

The flame as a source of sound is described by Klein et al. (1998). In the present paper the amplification by a flame will be considered.

Amplification by flames is a well-known phenomenon in combustion applications. It has already been studied for a long time.

With some simplifications the acoustic system of a combustion installation can be described with a control loop, see figure 1. In the frequency domain this control loop consists of three transfer functions.

In the case of gaseous turbulent flames it can be assumed that the main influence of the acoustic pressure on the flame behavior is via the fluctuating mass flow through the burner mouth, see for example Hermann (1997). These mass flow fluctuations are transported to the flame interface by the mean flow, where they generate a fluctuating heat release and add (or distract) energy to (from) the acoustic system.

Acoustic system description

The transfer functions for the three different components are:

I. **The combustion chamber transfer function** ($\tilde{T}_{cc}$). The transfer function for the combustion chamber can be derived from the equations for the propagation and reflection of sound. A typical shape for the combustion chamber transfer function around the eigen frequency of the combustion chamber (ω_n) can be derived from equation (1):

$$\tilde{T}_{cc}(\omega) \propto \frac{1}{\left(\omega^2 - \omega_n^2\right)} i\omega\tilde{Q} \tag{2}$$

II. **The burner transfer function** ($\tilde{T}_{bur}$). The burner transfer function gives the mass flow fluctuation (velocity fluctuation) in the burner mouth as a function of the pressure in the burner. For simple burners this transfer function can be calculated from acoustic equations. For more complex burners (cold flow) measurements are necessary. A typical shape of the burner transfer function is (if the pressure drop is low and the relationship between pressure and velocity fluctuations is pure acoustic):

$$\tilde{T}_{burner}(\omega) = \frac{\tilde{u}_{bur}(\omega)}{\tilde{p}_{bur}(\omega)} \propto \frac{1}{\rho_{bur} c_{bur}} \tag{3}$$

III. **The flame transfer function** ($\tilde{T}_{flame}$). The flame transfer function is the most important one because it may add energy to the acoustic system. The flame transfer function is defined by:

$$\tilde{T}_{flame}(\omega) = \frac{\gamma-1}{c^2}\frac{\tilde{Q}(\omega)}{\tilde{u}_{bur}(\omega)} \propto \frac{\gamma-1}{c^2}\frac{|Q_{total}|}{|U_{bur}|} e^{-i\omega\tau} \tag{4}$$

With τ the time delay necessary for a fluctuation to travel from the burner mouth to the flame interface.

Differences between premixed natural gas flames and non-premixed syngas flames

An important difference between natural gas flames, both premixed and non-premixed, and low calorific syngas flames is that in general in natural gas flames the area of the fuel injectors is much smaller and that the speed of the fuel relative to the air speed is much higher. This means that in general the pressure drop across a natural gas injector is much higher than the pressure drop across a syngas injector. This is due to the fact that the heating value of syngas is much lower than that of natural gas, so to keep the same thermal power the mass flow of syngas should be much higher. To be able to supply the high mass flow of syngas without too many losses the pressure drop across the syngas injector should be low. Because of this low pressure drop the syngas mass flow will be much more sensitive to pressure

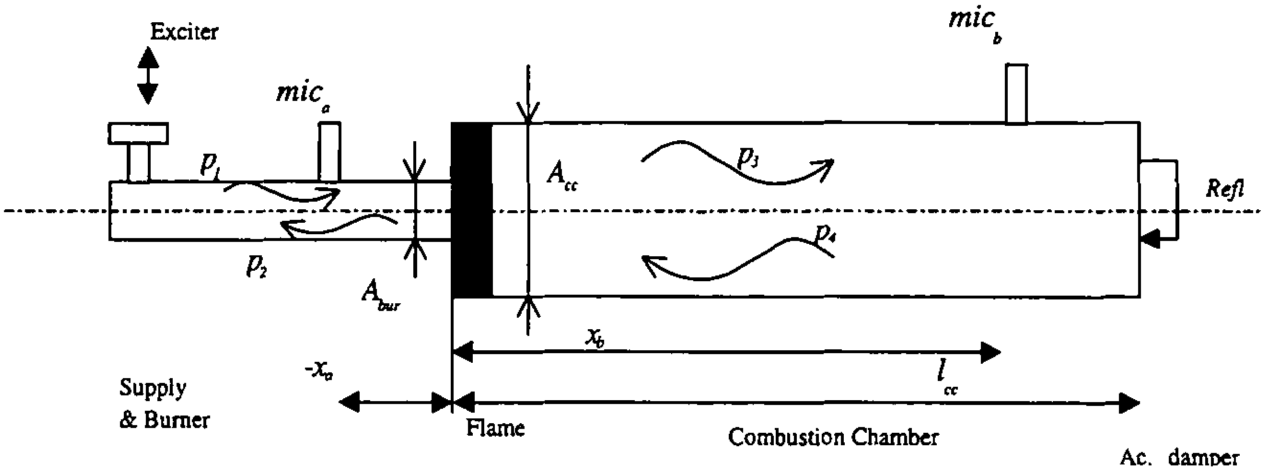

Figure 2: The experimental set-up

fluctuations in the combustion chamber than the mass flow of natural gas.

In a premixed natural gas burner the main fluctuations are in the airflow because of the high pressure drop across the fuel injector. These fluctuations in the airflow will induce fuel concentration fluctuations. After transport to the flame front these fuel concentration fluctuations will result in a fluctuating flame temperature and a fluctuating chemical reaction rate.

In a non-premixed flame the main part of the heat release takes place at the stoichiometric interface, where the fuel concentration and flame temperature are constant. This means that the chemical reaction rate and the heat release are also more or less constant at this interface. Fluctuations in the airflow will also only influence the total heat release by the flame if they change the flame shape strongly. Fuel fluctuations result directly, after transport to the flame interface, in a fluctuating total heat release, by changing the total flame volume. Non-premixed flames are also much more sensitive for fuel flow fluctuations than for airflow fluctuations. This has been tested in separate CFD calculations.

The weak sensitivity for airflow fluctuations of a non-premixed flame explains why non-premixed natural gas flames often show a very stable behavior. For a syngas burner the pressure drop across the fuel injector and the air injector are of comparable magnitude, so the fluctuations in the fuel flow will have the largest influence on the flame transfer function. In this study only this effect is described.

Rayleigh's criterion

From the wave equation a very simple criterion can be derived that states if a combustion system is stable or unstable, the Rayleigh criterion. This criterion says:

1. A flame amplifies the acoustic pressure in the combustion chamber if the phase between the acoustic pressure and the heat release is between minus ninety and ninety degrees:

$$-90^o < \varphi_{pq} < 90^o$$

2. Instability occurs if the amplification by the flame is larger than the total acoustic damping in the system.

The phase between the heat release and the pressure indicates the possibility for combustion instabilities. The amplification factor, in combination with the total acoustic damping, tells if they really occur.

EXPERIMENTAL METHOD AND SETUP

It is not possible to measure the heat release by the flame directly. In some experimental set ups use is made of the fact that the heat release by the flame is related to the amount of chemiluminescence of OH or CH radicals. For methane there exist a direct correlation between the heat release and the concentration of these radicals, for syngas there is not such a correlation. In the present experimental set up it is therefore chosen to measure the acoustic response of the flame and to calculate the heat release by the flame with an acoustic model. The measured acoustic quantities are the input for this model, the fluctuating heat release (as a function of the velocity fluctuation in the inlet) is the outcome of the model.

This experimental method is similar to the method used at ABB (Paschereit (1998)).

The acoustic method

To do acoustic measurements it is necessary to make the acoustic system of the combustion rig as simple as possible. This is done by making the maximum diameter of the system so small, that it can acoustically be treated as 1D. Because of this 1D behavior the propagation of sound can be described with plane waves.

The boundary conditions of the acoustic system should be well defined. The boundary conditions upstream of the membrane location, used to impose a fluctuation in the fuel flow, and in the air inlet of the burner are defined by creating a very high pressure drop. In this way it is assured that the mass flow through these flow restrictions does not fluctuate with the pressure in the combustion chamber. The air volume in the burner (see figure 3) has an acoustic eigen frequency (340 Hz) that is much higher than the frequencies studies in this research (<200 Hz), this will also not influence the measurements. The acoustic behavior of the fuel volume of the burner is taken into account in the acoustic model. The boundary condition at the exit of the combustion chamber is well defined by the use of an acoustic damper. This damper is a metal plate with a small hole. Its reflection factor is measured in cold flow measurements and is almost zero. The chamber with this damper can acoustically be treated as infinitely long.

The acoustic system of the combustion rig can be split into three parts (see figure 2): 1. the burner and supply tube, 2. the flame and 3. the combustion chamber.

An acoustic oscillation is imposed on the fuel flow by a membrane that is placed perpendicular to the flow. An exciter drives this membrane. The membrane generates sound. As a result of this sound the waves p_1, p_2, p_3 and p_4 will arise. The transfer function between the pressure measured with microphone mic_a and mic_b is determined with a FFT analyzer ($\tilde{H}_{meas}$). It is possible to write the waves $\tilde{p}_2$, $\tilde{p}_3$ and $\tilde{p}_4$ in the frequency domain as a transfer function times $\tilde{p}_1$. These transfer functions are determined by the conditions at the burner/flame, the reflection conditions at the exit of the combustion chamber and the measured transfer function. These conditions result in a linear set of four equations with five unknowns ($\tilde{p}_1$, $\tilde{p}_2$, $\tilde{p}_3$ and $\tilde{p}_4$ and $\tilde{T}_{flame}$). This set of equations is solvable because $\tilde{p}_2$, $\tilde{p}_3$ and $\tilde{p}_4$ can be expressed as a transfer function ($\tilde{b}_2$, $\tilde{b}_3$ and $\tilde{b}_4$) times $\tilde{p}_1$. This set of equations is (all equations are divided by $\tilde{p}_1$):

1.
$$1+\tilde{b}_2(\omega)=\tilde{b}_3(\omega)+\tilde{b}_4(\omega) \tag{5}$$

(The pressure is assumed to be continuous across the flame)

2.
$$\tilde{H}_{meas}\left(e^{-ik_{bur}x_a}+\tilde{b}_2(\omega)e^{ik_{bur}x_a}\right)= \left(\tilde{b}_3(\omega)e^{-ik_{cc}x_b}+\tilde{b}_4(\omega)e^{ik_{cc}x_b}\right) \tag{6}$$

($\tilde{H}_{meas}$ is the measured transfer function)

3.
$$\tilde{b}_4(\omega)e^{ik_{cc}l_{cc}}=Refl(\omega)\cdot\tilde{b}_3(\omega)e^{-ik_{cc}l_{cc}} \tag{7}$$

(*Refl* is the reflection factor at the outlet of the combustion chamber, known from cold flow measurements)

4.
$$-\left(ik_{bur}A_{bur}\left(\tilde{b}_2(\omega)-1\right)-ik_{cc}A_{cc}\left(\tilde{b}_4(\omega)-\tilde{b}_3(\omega)\right)\right)= -\frac{\gamma-1}{c_{cc}^{\;2}}i\omega\tilde{Q}(\omega)=-i\omega\cdot\tilde{T}_{flame}(\omega)\cdot-\frac{\tilde{b}_2(\omega)-1}{\rho_{bur}c_{bur}} \tag{8}$$

(The heat release by the flame is written as a function of the velocity fluctuations in the burner mouth. $\tilde{T}_{flame}$ is the flame transfer function)

The main source of possible errors is the determination of the reflection factor at the exit of the combustion chamber. In future measurements more microphones will be used in the combustion chamber, it is then not necessary to know this reflection factor in advance.

The combustion installation

The combustion installation, in which the experiments have been performed, is designed to study low calorific syngas flames.

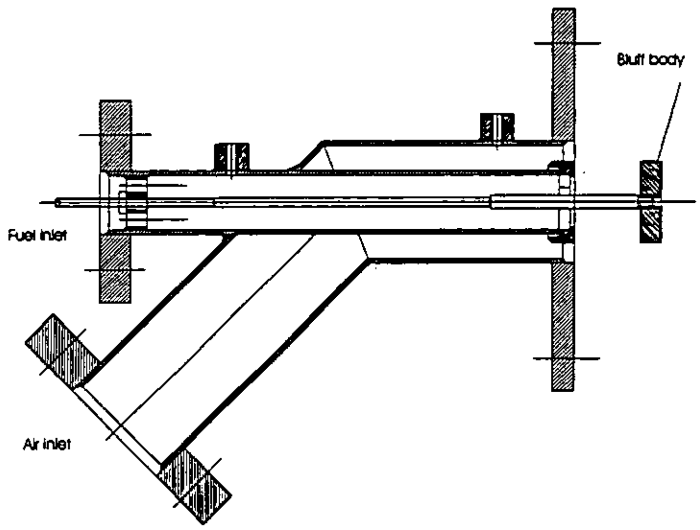

Figure 3: The non-premixed burner

The fuel supply in the laboratory can deliver fuels composed of methane, carbon monoxide, hydrogen, nitrogen and steam up to a thermal power of 100 kW. The fuel gasses, CO, H_2, N_2 and CH_4 are stored in separate cylinders. The mass flows of these components are controlled with mass flow controllers. After the mass flow controllers the gas flows of the different components are mixed. The fuel gas can be preheated up to 450 K. After the preheater steam can be added to the fuel gas.

The combustion air is supplied from a pressurized air system. The mass flow is controlled with a sonic nozzle. The airflow can be preheated up to 400 K.

The burner is of the non-premixed type, see figure 3. The central tube supplies the fuel; the concentric outer tube supplies the air. The flame is stabilized on a bluff body that is positioned in the core of the fuel exit. This bluff body increases the mixing of fuel and air and creates a recirculation zone. The exit diameter of the fuel tube is 28 mm, the exit diameter of the air tube is 50 mm.

The burner is attached to a cylindrical combustion chamber with a length of 1m. This combustion chamber consists of two cylindrical tubes. The inner tube, which has a diameter of 100 mm, is the actual combustion chamber. Because of this small diameter the combustion chamber is acoustically 1D for frequencies below 1500 Hz (c=600 m/s). Between the inner and outer tube there is a flow of cooling air. Microphones are attached to the combustion chamber via small tubes.

The microphones used in the experimental set-up are of the piezo-resistive type, Endevco 8510B1. The signals from the microphones are amplified (Endevco 106 amplifier) and analyzed with an FFT-analyzer (DIFA APB 200-8).

SIMULATIONS

A model has been developed to calculate the flame transfer function by transient combustion CFD calculations. As defined in eq. (4) the flame transfer function is the transfer function between the velocity fluctuations in the inlet and the fluctuating heat release by the flame.

In the CFD calculations the inlet velocity of the fuel oscillates harmonically as a function of time at a specific frequency. The response of the flame to this time-dependent inlet boundary condition is calculated. After the computation has converged for a time step the total heat release by the flame is calculated. From the results of the calculations the flame transfer function for that frequency can be calculated. These CFD calculations are performed for several frequencies. (This is another approach as the method described in Bohn (1996), they imposed a step function in the inlet air velocity.)

As the flame studied in this research is a non-premixed flame, it is possible to use the conserved scalar approach. It can be assumed that the chemical time scales are much smaller than the acoustic and turbulent time scales, all species are then in chemical equilibrium. If the flame can be treated as adiabatic all species, the temperature and the density can be written as a function of only the mixture fraction (Bilger (1980)).

The boundary condition for the fuel inlet velocity is:

$$u_{fuel}(t) = U + \hat{u}\sin(\omega \cdot t) \tag{9}$$

(With $\hat{u}$ 10% of the mean flow velocity (U))

The heat release by the flame is modeled as in the standard mixed-is-burnt approach, this means that it is expressed as a factor times the scalar dissipation (Bilger (1980)). The equation for the total heat release by the flame is (see also Klein (1998)) for a complete description):

$$\frac{\gamma-1}{c^2}q(t) = \frac{\gamma-1}{c^2}\iiint_V q(\underline{x},t)dV = \frac{1}{2}c_{g2}\iiint_V \frac{\varepsilon}{k}g\frac{1}{T}\frac{d^2T}{df^2}(f,g)dV \tag{10}$$

Equation (10) assumes that the acoustic pressure is constant over the flame. This is possible if the flame is short compared to the acoustic wavelength (a compact source of sound).

The models used in the CFD-calculations are: k-epsilon model for turbulence modeling, local equilibrium PDF approach for the species, temperature and density.

The transport equations are solved with the commercial CFD-package CFX. The combustion model is the equilibrium version of the F.I.R.S. model, developed at the University of Twente. The complete model is described in Kok et al. (1998).

The boundary conditions are: fluctuating velocity at the fuel inlet, constant velocity at the air inlet. Acoustic effects are not taken into account, this means that the density is only a function of the mixture fraction and not of the pressure. In CFX this is known as 'weakly compressible'. With this treatment the exit boundary condition is a Neumann condition with a correction to remain total mass conservation.

The computational grid is a 2D axisymmetric grid that consists of 2500 cells.

The calculations are performed for several frequencies (10,20,..,200 Hz), per frequency 50 time steps are taken for 5 periods.

The average CPU time per frequency is about 2.5 hours, the total CPU time for 20 frequencies is 50 hours.

RESULTS

In figures 4 & 5 the results from the measured flame transfer function and the calculated flame transfer function are plotted. The plots give the results for two flames with the same fuel composition (38 vol% CO, 5 vol% H_2, 57 vol% N_2) and the same air factor (λ=1.4) but with a different thermal power. Figure 4 gives the amplification factor, figure 5 the phase as a function of the frequency.

For frequencies above 50 Hz the phase is very well predicted both for the 12.5 kW flame and the 25 kW flame (see figure 5). The typical

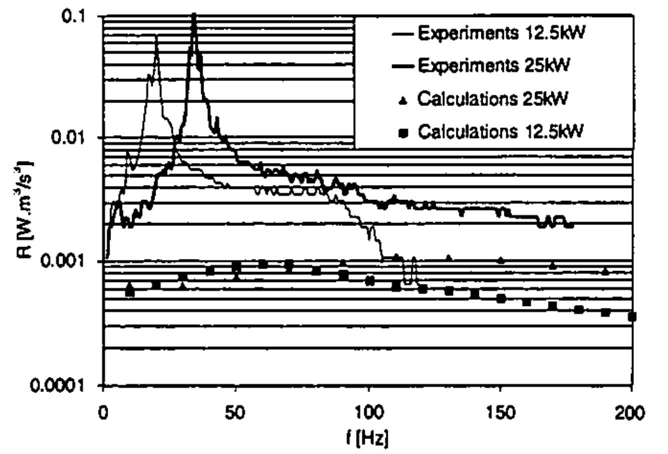

Figure 4: The amplification factor of the flame transfer function

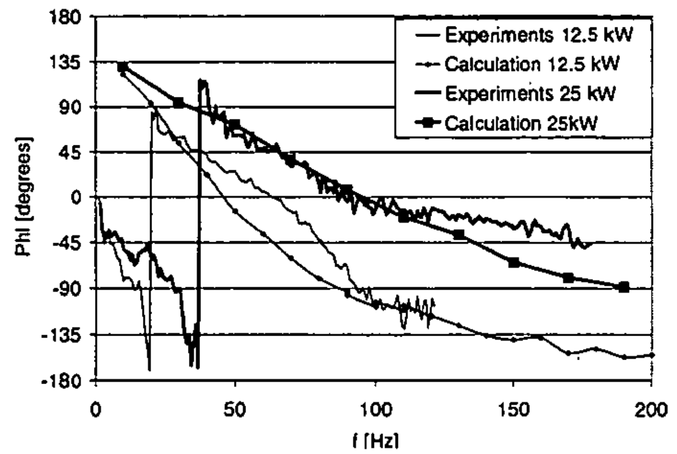

Figure 5: The phase of the flame transfer function

measured low frequency behavior of the phase with the jump from −180° to 135° is not predicted. This could be a consequence of the fact that the time steps for the low frequencies in the CFD calculations are too large. For higher frequencies the phase decreases linearly with the frequency. This indicates that there is a constant time delay for a fluctuation to travel from the burner mouth to the flame interface. The time delays (calculated from the slope of the phase in figure 5) for the 12.5 kW and the 25 kW flame are 8 ms and 4 ms respectively. As the mean flow speed of the 25kW flame is twice the flow speed of the 12.5 kW flame, this indicates that the average heat release position (≈ half the flame length) is the same for both flames.

The agreement between the measured amplification factor and the predicted one is less good as the agreement for the phase (see figure 4). For higher frequencies the calculated amplification factor of the transfer function is 4-8 times too small. This is mainly due to the use of a Reynolds (Favre) averaged turbulence model: In reality the instantaneous heat release takes place at concentrated locations where the turbulent eddies collapse, the positions of these locations fluctuate in time (see Poinsot (1987)). By the use of a RANS turbulence model phase averaging is applied to these concentrated locations, this means that the heat release will be smeared out. The integral in equation (10) is smaller for a distributed heat release than for a concentrated heat release, the amplification factor will also be smaller. The large peak that appears in the measurements of the amplification factor is not predicted. This peak occurs at the same frequency as the jump in the phase diagram. This peak is probably caused by flow instabilities, this assumption is supported by the fact that the frequency of the peak doubles by doubling the power.

In an accompanying experiment the instability frequencies have been determined. These measured instability frequencies show very good comparison with calculated ones from a simple acoustic model in which the flame is modeled by the measured flame transfer function.

CONCLUSIONS

It seems to be possible to separate the flame from the other components in the acoustic system of a combustion installation. The dynamic behavior of the flame can then be studied with CFD calculations.

The comparison between the measured flame transfer function and the calculated one is good. The flame transfer function shows the expected phase behavior (time delay) and above 50 Hz the amplification factor has the right order of magnitude.

It is not possible to calculate all the properties of the flame transfer function with the present CFD method. Especially flow instabilities and transient turbulence behavior have a significant effect on the amplification factor of the flame transfer function. It is not possible to capture this with a k-epsilon turbulence model. To improve the prediction of this aspect large eddy simulations will be necessary. At present a suitable L.E.S. model is implemented and being tested.

The CFD method is applicable in the process of burner design. This means that it is possible to perform a parameter study of the flame transfer function. With this parameter study it is possible to compare different burner designs. At the moment this is however very time consuming. Efficient algorithms have to be developed to overcome this problem.

If the flame transfer function is known it can be placed in a model of the acoustic system of a gas turbine combustion chamber. The fulfillment of the Rayleigh's criterion can easily be checked. If it is also possible to give a good estimate for the total damping in the acoustic system, it should be possible to give a prediction if the gas turbine will show combustion instabilities or not.

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